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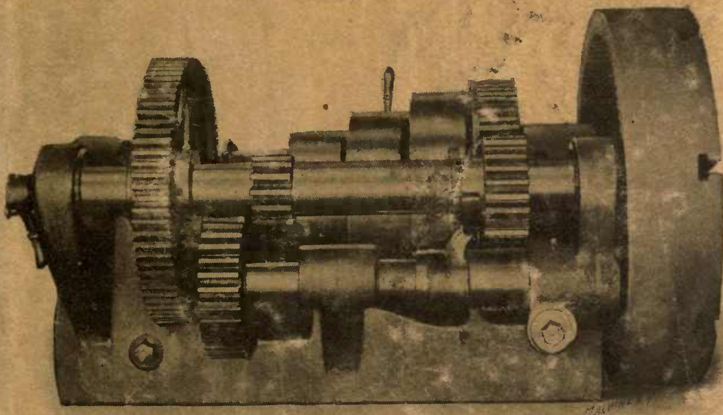
MACHINE TOOL DRIVES

SPEED AND FEED CHANGES—CONE PULLEYS AND GEAR RATIOS—SINGLE PULLEY DRIVES

THIRD REVISED AND ENLARGED EDITION



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CHAPTER I

DATA FOR THE DESIGN OF DRIVING AND FEED MECHANISMS

There is probably no branch of machine design in which greater changes have taken place in recent years than that of the design of machine tools. The greater part of these changes are without doubt due to the work of Mr. Fred W. Taylor, the discoverer of high-speed steel, who has more thoroughly investigated the capabilities and possible performances of metal cutting tools than any other man. The writer had occasion some time ago to study carefully Mr. Taylor's paper "On the Art of Cutting Metals." His study of this paper, together with his own experience in machine tool design and operation, has brought him to certain conclusions in regard to some points in machine tool design which will be of interest and value not only to those who may themselves design and build such tools, but also to everyone who has to purchase or use them.

Ratio of Speed Changes

The first point to which the writer would call attention is the necessity of a sufficient number of speed changes. Those who have read Mr. Taylor's paper will remember that he shows that there is a definite relation between the cutting speed and the length of time which a tool will last without regrinding. Should the machine be run at too high a speed, the tool will last but a short time before it will have to be reground. Should it be run at too low a speed, less work, of course, will be done, although the tool will last a comparatively long time. Somewhere there is a golden mean at which the cost of machining plus the cost of tool dressing is a minimum, and theoretically our machine should always be run at that speed. Of course, in handling materials of varying grades of hardness, and, in the case of lathes and boring mills, of varying diameters, this would necessitate a very great number of speed changes. If the number of speed changes be limited, it is apparent that the machine cannot always be working at the point of maximum efficiency. The speed of cutting which gives the maximum efficiency is shown in Mr. Taylor's paper to be that speed which will destroy the tool in from 50 minutes in the case of a $\frac{5}{8}$-inch $\times$ 1-inch roughing tool, to 1 hour and 50 minutes in the case of a 2-inch $\times$ 3-inch roughing tool. These times are of course only approximations and will vary somewhat with the cost of steel and labor and the value of the machine in which the tool is used. If the machine be slowed down from this proper speed, the cost of machining will slowly increase, but if the machine be speeded up above this proper speed, the cost of machining will increase very rapidly. In his paper Mr. Taylor gives a diagram wherein it is shown that if the machine be slowed down so that the duration of the cut is increased from 50 minutes to about 4

hours and 40 minutes, the machine is then working at about 90 per cent of its former efficiency. If the machine be speeded up until the duration of the cut is decreased to about 15 minutes, the machine will again be working at about 90 per cent efficiency. This range of speed is shown by Mr. Taylor's equations to be in the ratio of $\sqrt[8]{15}$ to $\sqrt[8]{280}$ or of 1 to 1.45. Consequently, if we have a machine having several speeds with the constant ratio of 1.45 between the successive speeds, we know that such a machine may always be made to operate within 90 per cent of its maximum efficiency, and that on the average it will operate at more than 95 per cent of its best efficiency.

The following table, which is derived in the manner indicated from the diagram given in Mr. Taylor's paper, shows the speed ratios corresponding to the given average and minimum efficiencies of working.

Ratio	Average Efficiency	Minimum Efficiency
1.1	99.6 per cent	99.2 per cent
1.2	98.7 per cent	97.3 per cent
1.3	97.3 per cent	94.5 per cent
1.4	95.6 per cent	91.2 per cent
1.5	93.5 per cent	87.0 per cent
1.6	90.6 per cent	81.2 per cent
1.7	86.5 per cent	73.0 per cent

From the table it will appear that even in the case of very costly machines it is of no particular advantage to reduce the ratio between successive speeds unduly. For instance, by doubling the number of speeds and reducing the speed ratio from 1.2 to 1.1, we will increase the average efficiency of the machine only about 1 per cent. It is very doubtful if the accidental variations in shop conditions would not be so great that the gain in practical work would be nothing, since the workman or the speed boss, as the case might be, would be unable to decide which of two or three speeds would be the best. The writer is therefore of the opinion that there is absolutely no practical advantage in reducing the speed ratio below 1.2 and that in the case of machines of ordinary type and cost, a ratio of 1.3 is as small as is advisable. In the case of a speed ratio of 1.3, the machine can always be made to operate at such a speed that the efficiency of working will be above 94.5 per cent and in the average case the efficiency will exceed 97.5 per cent. The 2.5 per cent loss of efficiency so caused is inappreciable as compared with other sources of loss, and it is exceedingly doubtful if the added cost of additional speed changes would not more than compensate for the possible 1 or 2 per cent of gain, entirely aside from the question of whether the extra speed changes would permit this theoretical gain to be realized.

The writer is also of the opinion that a speed ratio of more than 1.5 in the case of expensive machinery operated by highly skilled help, or of 1.7 in the case of cheap machinery operated by comparatively unskilled help is inadvisable. It will be seen that with a speed ratio of 1.5 the average efficiency of working is, in general, about 93.5 per cent, making the loss of efficiency in the average case about 6.5, or say 6 per cent. It will be seen that when the rent of the tool plus the wages of a mechanic amounts to $4 a day or upward, this 6 per cent of loss means

a money loss of $0.25 or more per day, or upward of $75 a year. Of course, an increase in the number of speed changes and reduction of ratio would not save all this loss, but assuming that it would save half of it, and further, that the machine is operating only half the time, it is evident that we can afford to spend $150 or $200 for the extra speed changes necessary in order to bring the speed ratio down to 1.3. In the case of a ratio of 1.7, the loss is 12 or 13 per cent instead of only 6 per cent, and these figures apply with greatly added force.

We are thus compelled to the conclusion that the useful range of the speed ratio in machine tool work is very narrow, ranging from 1.3 to 1.5 in ordinary cases and that a range of from 1.2 to 1.7 includes the very extremes of rational practice.

Need of Speed Changes Being Easily Made

A second point in connection with the matter of the speed changes of machine tools which is of great importance is that these changes should be easily and quickly made so that the operator will have every incentive to use the proper speed. This is a matter of less importance in the case of planers than in the case of lathes and boring mills, since a planer requires a change of speed only when the character of the material which is being cut is changed, while the lathe requires a change when any great change is made in the diameter of the work operated upon.

In this respect a motor-driven tool may have a distinct advantage over a belt-driven tool. The controller furnishes a ready means for varying the speed while the shifting of a belt from pulley to pulley is not always readily accomplished, and most machinists would much rather take two cuts of differing diameters on the back-gear than shift the belt from the small to the large pulley and throw out the back-gear in order to obtain the faster speed from the open belt. This is particularly the case when the cuts are of small duration, so that the shifting would be frequent.

It will be evident to the thoughtful mechanic that it is of great advantage to have the speed-changing mechanism so constructed that the change may be made without stopping the machine. In the case of large machines it will be of great advantage to be able to effect the speed change from the operating station, which for instance in the case of a long lathe will be the carriage. To the writer's mind the particular advantage of these refinements which he suggests, and which will be found embodied in many of the designs of our best tool makers, lies not in the fact that the time required to make the necessary speed changes is shortened, but in the fact that the workman finds it just as easy to run his machine at the proper speed as at an improper one.

Ratio of Feed Changes

A matter of even greater importance than a proper series of easily made speed changes is a proper series of easily made feed changes. A change of speed does not mean in general a correspondingly great change in the efficiency of operation of a machine tool, but a change in feed does. Mr. Taylor points out in his paper that in general the best

results in quantity of metal removed per hour are obtained when the cross-section of the chip is a maximum, even though this entails a comparatively low speed. Therefore it is of importance that the machinist be able to take the heaviest cut which the nature of his work and the power and stiffness of his machine will permit. Just as the best results in the matter of cutting speeds are obtained when the successive speeds run in geometric ratio, so the best results in the matter of feed adjustment are obtained when the successive feeds run in geometric ratio, unless the number of obtainable feeds is so great that the entire range is closely covered. For instance, a lathe equipped with the following feeds, 0.05, 0.10, 0.15, 0.20, 0.25, is distinctly inferior in productive capacity to a lathe having the same number of feeds arranged geometrically as follows, 0.05, 0.074, 0.111, 0.166, 0.25, wherein each feed is about 50 per cent greater than the preceding one.

In general the best work is obtained from a machine tool when the depth of cut is made such that the total depth of metal to be cut away is removed with one or two cuts. Such being the case, the depth of cut is practically fixed and not within the control of the operator, leaving the feed and speed as the variables which he must adjust. It is important therefore that the operator be able to take a cut as heavy as the nature of the work or of the tool will permit. Mr. Taylor's paper shows that the speed of cutting is approximately inversely proportional to the square root of the feed. It needs therefore only a very elementary knowledge of mathematics to see that if the feed must be reduced to say 80 per cent of its maximum value, the output of the lathe will be only about 90 per cent of its maximum value. Or in general, if the feed be reduced from its maximum possible value by any given per cent, then the output of the machine will be reduced from its corresponding maximum value by about one-half of that per cent. We may by means of this principle compute the ratio between successive feeds which will give us any required average value for the efficiency of operation of the machine. The values so found are tabulated below:

Efficiency	Ratio	Efficiency	Ratio
98 per cent	1.08	90 per cent	1.66
96 " "	1.18	88 " "	1.92
94 " "	1.32	86 " "	2.27
92 " "	1.46		

An inspection of the table shows that when the ratio between successive feeds is about 1.1, the average efficiency of operation of the machine may be practically perfect, and that with any considerable increase of this ratio the efficiency drops off. It is the opinion of the writer that the ratio between successive feeds should always be less than 1.3 and that, more especially in the case of expensive machinery, a value of 1.2 or less is preferable.

Importance of Convenience of Feed-changing Mechanism

It has already been pointed out that the speed-changing mechanism should be of such a character that the speed changes may be easily and quickly made. In the same way it is of even greater importance that the feed changes may be easily and quickly made. In most small

lathes which are now on the market, quick-change gears are fitted to the screw-cutting mechanism, which are equally available as quick-change gears for the feed mechanism. In most shops small lathes are not used very much of the time for screw-cutting, and in fact nine lathes out of ten are never used for that purpose, but a quick-change gear mechanism is of much greater importance when used for the purpose of obtaining feed changes than when used for the purpose of obtaining thread changes. In the average case the operator will not have to touch the thread-cutting gear once a week, while it may be advisable to change the feed every five minutes. In the case of large lathes it is advisable to have the feed changes, not in the head-stock but in the apron, in order that the workman may be encouraged to use a proper feed whenever possible.

Unlike lathes, planers are generally equipped with ratchet feeds. The successive values of the feed changes in the case of a ratchet feed will necessarily run in an arithmetic and not a geometric series, the successive feeds differing by some constant decimal of an inch. So long as the amount by which the successive feeds differ is small, and the range of feeds given by the mechanism is large, a ratchet feed is perfectly satisfactory. Many boring mills are fitted with a feed mechanism driven by a friction wheel of the type generally known as a brush wheel, the driving mechanism consisting of a steel disk of 12 to 16 inches in diameter geared to the table, and against the face of which a much smaller wheel edged with leather is pressed. It is obvious that if the steel disk rotate at a constant speed, the speed of the driven wheel and consequently the amount of the feed may be varied by adjusting its position. When it presses the disk near its center it will revolve slowly. When it presses the disk near its edge, it will revolve at a comparatively high speed. This feed mechanism has the advantage that it gives an infinite number of feed changes over a wide range, but has the disadvantage that it is not positive in its action, and lacks sufficient power for certain kinds of work. On the whole, the best feed driving mechanism is a nest of gears so arranged that any feed within the entire range may be had by the simple shifting of one or two levers.

Strength of the Feed Mechanism

In that part of his paper discussing the force required to feed the tool of a lathe or boring mill, Mr. Taylor makes the assertion that the feed mechanism should have sufficient strength to "deliver at the nose of the tool a feeding pressure equal to the entire driving pressure of the chip upon the lip surface of the tool." This would lead to the designing of a lathe or boring mill having feed gearing of equal strength with its driving mechanism. In the case of planers and other machines in which the tool is moved at a time when it is not cutting, these statements do not apply. It is not generally the custom among machine tool builders to design machines having such strong feed works as Mr. Taylor's ideas call for, and the writer sees no reason why such strength is necessary. The amount of force required to traverse a tool in a lathe is not proportional to the width of feed,

and while it may be true for fine feeds that in the case of dull tools the traversing pressure may be equal to, or greater than the downward pressure upon the tool, this is not necessarily the case with heavy feeds. As the width of the feed is increased, the downward pressure will increase almost in proportion, while the traversing pressure will increase comparatively little, so that when the lathe is taking the maximum cut which the driving mechanism is capable of handling, the pressure required to feed the tool into the work, even though it be very dull, is much less than the downward pressure. It is the writer's opinion that a feed mechanism designed to have one-half the strength of the driving mechanism is ample for large tools, while for small tools in which of course the feed will be finer, a strength of two-thirds of the driving mechanism might be preferable.

"Breaking Piece" of Feed Mechanism

The feed mechanism should be provided with a breaking piece whose strength will be less than that of the rest of the mechanism and which may be cheaply and easily replaced. The office of this piece is to prevent the breaking of the more costly and less easily replaced parts of the mechanism, exactly as the fuse in an electric circuit prevents the destruction of any other part of the circuit. Two forms of breaking piece sometimes used for such service are, first, a soft steel pin, driven through a shaft and hub of harder steel, which shears off when the strain becomes too great; and second, a short section of shaft turned down at its center, which twists off under similar circumstances. A breaking piece must be of such a character that it will not spoil any of the rest of the mechanism when it breaks, and should not cost more than a few cents, and should be as easily removed and replaced as a common change gear.

It must not be imagined that a feed gearing designed to have one-half the strength of the driving gear will not be strong enough to meet Mr. Taylor's requirements in all ordinary cases. If a tool be designed to take a maximum cut of 3/8 inch by 1/8 inch, it is not likely that much of its work will be done with such a heavy cut. If both driving and feed gearing be designed with a proper factor of safety, there is ample margin of strength for all usual conditions, while a breaking piece is the best provision against extraordinary stresses.

Pressure on Lip Surface of Tool and Its Relation to Design

The pressure upon the lip surface of the tool is required in order that the designer may know, first, the strength required of the driving mechanism and frame of a machine; second, the power required by the machine; and third, the strength required for the feed mechanism. The two materials upon which the vast majority of machine tools are called to operate are cast iron and steel. Taking first the case of cast iron, we find from Mr. Taylor's paper that the pressure upon the lip surface of the tool varies from 75,000 to 150,000 pounds per square inch of chip section in the case of soft iron, and from 120,000 to 225,000 pounds in the case of hard cast iron. The finer the feed, the greater the pressure per square inch upon the lip surface of the tool.

Thus with an ⅛-inch depth of cut and 1/64-inch feed, the pressure on the tool is about 289 pounds, or 146,000 pounds per square inch. With the same depth of cut and ⅛-inch feed, the pressure on the tool is 1,358 pounds, or only about 86,900 pounds per square inch of chip section. Both these figures are given for soft cast iron. Mr. Taylor gives formulas for the total pressure of the work upon the lip surface of the tool, but the following table will be found more convenient for obtaining the required values, although the figures given are of course only approximations:

	Pressure per Square Inch	
Feed, Inches	Soft Cast Iron	Hard Cast Iron
1/64	140,000	220,000
1/32	120,000	190,000
1/16	100,000	160,000
1/8	85,000	135,000

In the case of soft and medium steels we find that the pressure in pounds per square inch of chip section runs from 250,000 to 300,000 pounds, being greater in the case of the finer feeds. In the case of special steels which combine high tensile strength and great elongation, it is probable that these figures would be very much exceeded. The amount of the feed and depth of cut will depend on the kind of work which is to be machined. In the case of small castings, 3/16 inch is an ample allowance for depth of cut and ⅛ inch would be much more usual. In the case of very large and heavy castings the depth of cut required might run up to ½ inch, and in the case of large "meaty" forgings, it may be even greater than this at some places. In those cases where the area of chip section is not fixed by the work, as in the case of stocky forgings and castings, the greatest width of feed is limited by the strength of the machine itself, which in turn is limited only by the length of the purchaser's purse. Presumably it would be possible to build a boring mill or a planer capable of taking a cut an inch deep with an inch feed if anyone wished to pay for such a machine, but whether it could do the average line of work as economically as a machine taking a ⅜-inch cut with ⅛-inch width of feed is another matter. While there is no settled rule either for the maximum depth of cut or width of feed for any particular type of machine, the matter of the size of tool used is generally definitely known. In the case of forged roughing tools the maximum chip section will be from 2 to 3 per cent of the area of the section of the tool shank. For instance, the heaviest cut which a tool forged from 1-inch by 1½-inch stock will be called upon to take will be ¼ inch by ⅛ inch, or perhaps a trifle greater. In the case of tools ground from bar stock and held in tool-holders, the section of the chip may run up as high as 5 per cent of the section of the bar. Knowing the size of tool for which the tool-holders are designed, we may proportion our machine accordingly.

A matter which has great effect not only upon the quantity of work which a machine is capable of doing, but also upon its accuracy and length of useful life, is its stiffness. While it is true that if we know the maximum pressure upon the lip-surface of the tool, we may design

a machine for strength and have one which will probably never break in service, yet it is often better to add many times the quantity of metal which mere strength would call for, in order to have a machine with the maximum of stiffness. Stiffness in machine tool design has to do with two points, the first being the actual deflection of the metal of which it is composed under the stresses which come upon it in operation; the second is the play which invariably exists at all joints, more especially the slides of compound rests in lathes, and of saddles in boring mills and planers. The best remedy for actual deflection of metal is to use plenty of it, and to distribute it in such a way as to realize from it its maximum strength. The writer has found that an excellent method of designing such machine parts as require great stiffness is by comparison with existing tools whose operation is satisfactory. Let us assume for instance that we are to design the cross-rail of a planer. The rail is to be 8 feet between the housings and the overhang of the tool below the center of the rail is to be 30 inches. The cut is to be, let us say, ½ inch deep by ⅛ inch feed. Let us assume further that we have at our disposal a 4-foot planer, the overhang of whose cutting tool is 15 inches, and which will take in a satisfactory manner a cut ¼ inch deep by 1/16 inch feed. We now have sufficient data to satisfactorily design a cross-rail for the larger planer. If we assume that the deflection of the tool produced in the two cases should be identical in order to have the work equally satisfactory, we will find that the pressure upon the tool of the larger planer will be 4 times that upon the tool of the smaller; that both the bending and the twisting moments set up in the cross-rail will be 8 times as large, and that the distance over which these moments will operate to produce a deflection will be twice as great. Therefore, if the two rails had the same cross-section, the deflection of the tool of the larger machine would be 16 times that of the tool of the smaller. The stiffness of two bodies of similar section varies directly as the 4th power of the ratio of their homologous dimensions. Therefore, if we make the section of the rail of the larger machine similar in form to that of the rail of the smaller machine, each dimension twice as great as the corresponding dimension of the smaller rail, it will be 16 times as stiff and the deflections in the two cases will be identical. In case the rail of the smaller machine were not of the best form to resist the stresses which it must sustain, the form might be changed, the designer using his best judgment as to what effect such change might have upon its stiffness.

CHAPTER II

SPEEDS AND FEEDS OF MACHINE TOOLS

In designing machine tools of any type, be it a lathe, milling machine, grinding machine, etc., aside from the correct proportioning of the parts, and the introduction of convenient means for rapidly producing certain motions, a very important factor is to be taken into consideration, that is, the correct proportioning of the speeds and feeds of these various machines. Before entering into an explanation of the method which is to be set forth later, we will explain some of the preliminary considerations which are to be met by the designer. Supposing a problem of designing a lathe be presented; it follows, at once, that certain conditions limiting the problem are also given. These limiting conditions may be considered as the size and material of the piece to be turned.

We consider the material of a piece to be machined as a limiting condition for the reason that a lathe turning wood must run at a different speed from one turning brass, and the latter at a different speed from a lathe turning iron or steel. Then, again, in turning a small piece, our machine will revolve faster than in turning a large piece. The speeds required for machining advantageously the different materials, according to the different diameters, may be termed "surface speeds." Roughly speaking, the surface speeds for the different materials vary within comparatively narrow limits. We may assume the following speeds for the following materials (using carbon steel cutting tools):

Cast iron	30 to 45 feet per minute.
Steel	20 to 25 feet per minute.
Wrought iron	30 feet per minute.
Brass	40 to 60 feet per minute.

For cast iron as found in Europe, we may assume 20 to 35 feet per minute; this lower figure is due to the fact that European cast iron is considerably harder.

The surface speeds above given are, of course, approximate, and it is left to the judgment of the designer to modify them according to the special given conditions. These surface speeds for cutting metal are the same whether the piece to be cut revolves, or the cutting tool revolves around the piece, or, as in a planer, the cutting tool moves in a straight line along or over the work. Therefore, the surface speeds in a general sense hold good for all types of machines, such as milling machines, lathes, gear-cutting machines, drilling machines, planers, etc.

Suppose that a problem is given requiring that a lathe be designed to turn both cast iron and steel, and to turn pieces from one-half inch

to twelve inches in diameter. Simple calculation will show us that a piece of work one-half inch in diameter, and having a surface speed of 30 feet per minute, as would be suitable for cast iron, must make 230 revolutions per minute. A piece of steel, which is 12 inches in diameter, with a surface speed of 20 feet per minute, must make 6.5 revolutions per minute approximately. It follows that the lathe to conform to the conditions imposed, must have speeds of the spindle varying from 6.5 to 230 revolutions per minute. These are the maximum and minimum speeds required. To meet the varying conditions of intermediate diameters, the lathe will be constructed to give a certain number of speeds. The lathe, probably, will be back-geared and have a four-, five-, or six-step cone.

In a correct design these various speeds must have a fixed relation to each other. For reasons explained in Chapter III these speeds must form a geometrical progression, and the problem briefly stated is this: "The speeds (the slowest and fastest being given) are to be proportioned in such a manner that they will form a geometrical progression." The ratio of the gearing is also to be found. A geometrical progression in a series of numbers is a progressive increase or decrease in each successive number by the same multiplier or divisor at each step, as 3, 9, 27, 81, etc.

To treat the problem algebraically let there be

$n =$ number of required speeds,
$a =$ slowest speed,
$b =$ fastest speed,
$d =$ number of speeds of cone,
$n-1 =$ number of stops or intervals in the progression of required speeds,
$f =$ ratio of geometrical progression, or factor wherewith to multiply any speed to get the next higher.

Algebraically expressed, the various speeds, therefore, form the following series:

$$a, af, af^2, af^3 \ldots\ldots af^{n-2}, af^{n-1}$$

The last, or fastest speed, is expressed by af^{n-1} and also by the letter b. Therefore, $af^{n-1} = b$, or

$$f^{n-1} = \frac{b}{a}, \text{ and } f = \sqrt[n-1]{\frac{b}{a}}$$

Suppose we have, as an example, a lathe with a four-speed cone, triple geared. In this case we would have four speeds for the cone, four more speeds for the cone with back-gears, and still four more speeds with triple-gears; therefore, in all, twelve speeds. Assuming a as the slowest speed in this case, b would be expressed by af^{11}, and the series, therefore, beginning with the fastest speed, would run

$$af^{11}, af^{10}, af^9 \ldots\ldots af^2, af, a.$$

The four fastest speeds, which are obtainable by means of the cone alone would be

$$af^{11}, af^{10}, af^9, af^8.$$

Dividing each of the four members of this series by f^4, we obtain the following series:

$$af^7,\ af^6,\ af^5,\ af^4,$$

as the speeds of cone with back-gears.

Again dividing the series of speeds of the cone af^{11} to af^8 by $f^4 \times f^4 = f^8$ we obtain the series

$$af^3,\ af^2,\ af,\ a,$$

as the series of speeds of cone with triple-gears.

We have, therefore, in this way accounted for all the twelve speeds that the combination given is capable of, and it is now very evident that the ratio of the back-gears must be f^4, or, in general, f^d, if $d =$ number of speeds of cone, and the ratio of triple-gears f^8 (or, in general, f^{2d}).

By carrying this example still further, we would find that the ratio of quadruple-gears would be f^{3d}.

We can summarize the preceding statements, and put them in a more convenient form for calculation by writing:

$$\begin{aligned} lg \text{ of ratio of back-gears} &= d\, lg\, f \\ lg \text{ of ratio of triple-gears} &= 2d\, lg\, f \\ lg \text{ of ratio of quadruple-gears} &= 3d\, lg\, f \end{aligned}$$

The problem, with this consideration, therefore, is solved. An example will be worked out below.

We will now consider a complication of the problem which very often occurs. Should the overhead work of the drive in consideration have two speeds, then we will obtain double the number of available speeds for the machine, and this number of speeds may be expressed by $2n$, in order to conform to the nomenclature used above. This modified problem is treated just as the problem above, and the series of speeds is found as in the first case, and we have as a factor

$$f = \sqrt[2n-1]{\frac{b}{a}}$$

We must consider now that one-half the obtained speeds are due to the first overhead speed, the other half to the second.

In writing the odd numbers of speeds found in one line, and the even numbers of speeds in another, we obtain the following two series:

$$a,\ af^2,\ af^4 \ldots\ldots\ldots\ldots af^{2n-4},\ af^{2n-2}$$
$$af,\ af^3,\ af^5 \ldots\ldots\ldots\ldots af^{2n-3},\ af^{2n-1}$$

In examining these two series, we will find that they are both geometrical progressions, and furthermore, that both progressions have the same factor, and calling this factor, f_1, we have

$$f_1 = f^2,$$

and the ratio of the two counter-shaft speeds is equal to f, because to obtain any speed in the second series we multiply the corresponding speed in the first series by f. The two series in our case are due to the two overhead speeds. We need to concern ourselves with only one (either one of the two series), and without going again through the

explanation for the first case, it is very evident that we will arrive at the following conclusions:

$$lg \text{ of ratio of back-gears} = d\, lg\, f_1$$
$$lg \text{ of ratio of triple-gears} = 2d\, lg\, f_1$$
$$lg \text{ of ratio of quadruple-gears} = 3d\, lg\, f_1$$

Having in this way obtained all the desired speeds and the ratios of the gears, it is a simple matter for the designer to determine the actual diameters of the various steps for the cone and for the gears. To do so he has at his disposal various methods,* which need not be explained here. The main thing for him to have is a geometrical progression of speeds, as a foundation for his design.

Problem 1. A Triple-Geared Lathe

Suppose the following example to be given: Proportion the speeds and find the gear ratio of a six-step cone, triple-geared lathe; slowest speed, 0.75 revolution per minute; fastest, 117 revolutions per minute.

This example of a six-step cone, triple-geared, will give us eighteen available speeds. Using our previous notation, $n = 18$, $n-1 = 17$, $a = 0.75$, and $b = 117$; therefore

$$f = \sqrt[17]{\frac{117}{0.75}} = \sqrt[17]{156}$$

The slowest speed being given, we multiply it by the factor f to obtain the next higher, and this one in turn is again multiplied by the

COMPLETE CALCULATION OF CONE PULLEY SPEEDS

lg 0.75 = 0.8750613 — 1	1.0361270 = lg 10.867
lg f = 0.1290073	0.1290073
0.0040686 = lg 1.009	1.1651343 = lg 14.626
0.1290073	0.1290073
0.1330759 = lg 1.358	1.2941416 = lg 19.685
0.1290073	0.1290073
0.2620832 = lg 1.828	1.4231489 = lg 26.494
0.1290073	0.1290073
0.3910905 = lg 2.461	1.5521562 = lg 35.658
0.1290073	0.1290073
0.5200978 = lg 3.312	1.6811635 = lg 47.991
0.1290073	0.1290073
0.6491051 = lg 4.457	1.8101708 = lg 64.591
0.1290073	0.1290073
0.7781124 = lg 5.999	1.9391781 = lg 86.932
0.1290073	0.1290073
0.9071197 = lg 8.074	2.0681854 = lg 117.000
0.1290073	

* See MACHINERY'S Reference Series, No. 14, Details of Machine Tool Design, Chapters I and II.

factor f, and so on, until we have reached the highest speed b. The 17th root of 156 is easiest found by the use of logarithms.

We have

$$lg\,156 = 2.1931246$$
$$lg\,f = 1/17\ lg\,156 = 0.1290073$$
$$f = 1.3459$$

Now we follow out the multiplication by finding the logarithm of 0.75, the slowest speed, adding to it the logarithm of the factor f to obtain the logarithm of the next higher speed; and adding the logarithm of factor f to the sum of these two logarithms will give us the logarithm of the next higher speed. By looking up the numbers for these logarithms, we find these speeds to be 1.009 and 1.358. The complete calculation is given in tabulated form on the previous page.

Now, for example, the number of speeds of cone d equals 6, and according to our formula, the logarithm of the ratio of the back-gears $= d\ lg\ f$, and the logarithm of the ratio of the triple-gears $=$ $2d\ lg\ f$. Expressed in figures we have:

$lg\,f = 0.1290073 \times 6 = 0.7740438$, and the ratio of the back-gears $=$ 5.9435. Further, 12 $lg\,f = 1.5480876$, and the ratio of the triple-gears $=$ 35.325.

Problem 2.—Lathe with two Counter-shaft Speeds

Suppose the following example is given: Proportion the speeds and find the gear-ratio of a four-step cone, back-geared, two speeds to counter-shaft; slowest speed, 25 revolutions per minute; fastest speed, 500 revolutions per minute.

In this case $n = 8$; $2\,n = 16$; and, consequently,

$$f = \sqrt[15]{\frac{500}{25}} = \sqrt[15]{20} = 1.221$$

In following out the calculation as shown in Problem 1, we obtain the following series of sixteen speeds:

1) 25.00	5) 55.58	9) 123.54	13) 274.64
2) 30.53	6) 67.86	10) 150.85	14) 335.35
3) 37.28	7) 82.86	11) 184.20	15) 409.48
4) 45.51	8) 101.18	12) 224.92	16) 500.00

Of these sixteen speeds, eight are due to one over-head work speed; the other eight are due to the second over-head work speed. We write the odd and even speeds in two series, as below:

First Series.	Second Series.
1) 25.00	2) 30.53
3) 37.28	4) 45.51
5) 55.58	6) 67.86
7) 82.86	8) 101.18
9) 123.54	10) 150.85
11) 184.20	12) 224.92
13) 274.64	14) 335.35
15) 409.48	16) 500.00

In order to find the ratio of the back-gears, we can use either one of these two series, and as explained above, $f_1 = f^2$. We therefore

have $1.221^2 = f_1$, and further $4 \times lg\, f_1 =$ ratio of back-gears. From this the ratio of the back-gears = 4.9418. We also know that the ratio of counter-shaft speeds $= f = 1.221$.

This method of geometrically proportioning speeds in machine drives, which has been explained at length, will be found, after one or two applications, a rather simple one. But its usefulness is not limited to the proportioning of speeds in machine drives, as it can also be applied to the proportioning of feeds.

Feeds for Machine Tools

Before proceeding to apply this method to geometrically proportioning feeds in machines, a few remarks on feeds may not be out of place. By feeds are understood the advances of table, carriage, or work, in relation to the revolutions of the machine spindle. Feeds may be expressed in inches per minute or inches per revolution of spindle. In a table given below, feeds for different machines are given in inches for one revolution per spindle, where not otherwise specified. This table is supposed to represent modern practice, with carbon steel cutting tools, but the figures given, of course, represent general experience, and special cases, no doubt, will often modify them considerably.

	Feed, Inches.
Plain milling machine	0.005 – 0.2
Large plain milling machine	0.010 – 0.3
Universal milling machine	0.003 – 0.2
Large universal milling machine	0.003 – 0.25
Automatic gear cutter, small	0.005 – 0.1
Drills (spindle-feed)	0.004 – 0.02
Planing machine (traverse feed)	0.005 – 0.7
Slotting machine (feed of work)	0.005 – 0.2
Drilling long holes in spindles (per revolution of drill)	0.003 – 0.01
Lathes, feed for roughing	56 – 80 turns per inch
Lathes, feed for finishing	112 turns per inch.

Universal Grinding Machine

Surface speed of emery-wheel, 4,000–7,000 feet per minute. Traverse of platen or wheel, 2 to 32 inches per minute; the fast feeds are for cast iron. Surface speed of work on centers, 130–160 feet per minute. For internal work use the following surface speeds of emery-wheel (highest nominal speeds), with no allowance for slip of belt; lowest nominal speed about 40 per cent less. Any speed between should be obtainable.

Diameter of Wheel.	Feet per Minute.
1 5/8	3,600
1	2,750
3/4	2,100
7/16	1,450
1/4	1,100

Surface Grinding Machine

Surface speed of emery wheel, 4,000–7,000 feet per minute. Table

speed per minute, 8–15 feet. Cross feed to one traverse of platen, 0.005–0.2 inch. Cross feed to one revolution of hand-wheel, 0.25 inch.

Problem 3—The Feeds of a Milling Machine

The problem of proportioning the feeds of different machines varies in each case, although always embodying similar principles. It is, therefore, proposed to take a typical case and apply the method to the problem presented, and in this way explain the advantages of the particular method referred to.

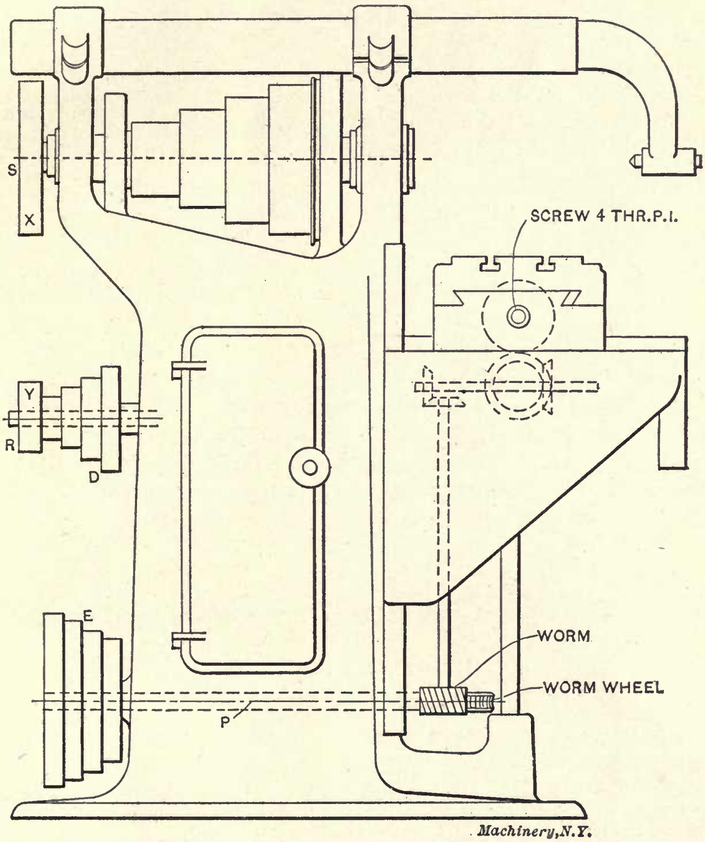


Fig. 1. General View of Milling Machine, having Cone Pulley Feed

In Fig. 1 is given an outline drawing of a milling machine. The type selected is not one of the latest designs, because it is easier to comprehend the principles involved in a type such as shown. The application of the principles, however, is, with few modifications, the same for the most modern gear-feed types, as for the one shown. The problem in this case will be the following: Given the fastest and slowest feeds per one revolution of main spindle, proportion the

required feeds in such a manner that they will form a geometrical progression. Cones D and E as well as pulleys X and Y can be transposed.

The main data with which we have to concern ourselves about this machine may be assumed to be as follows: lead screw, four threads per inch, single; advance of screw per one revolution, 0.25 inch; largest feed wanted, 0.25 (equal to one revolution of screw); smallest feed wanted, 0.005 inch (equal to 1/50 revolution of screw); for one revolution of screw, shaft P (see Fig. 1) makes thirty revolutions; for 1/50 revolution of screw, shaft P makes $30 \div 50 = 0.6$ revolutions. The ratio of revolutions between the screw and shaft P is therefore in our example as 1 to 30; that is, given the revolutions of shaft P we divide this number by 30 to obtain the revolutions of the screw. The revolutions of the screw multiplied by the lead L (in this case equal to 0.25) gives the advance for given revolutions of P. Let

V = ratio of train from P to screw,

L = lead of screw,

R_p = revolutions of shaft P per one revolution of spindle,

p = advance or feed of screw per one revolution of spindle, expressed in inches.

We have

$$p = \frac{R_p L}{V} \qquad (1)$$

$$R_p = \frac{Vp}{L} \qquad (2)$$

If now n equals the numbers of feeds wanted, we obtain for f, the factor wherewith to multiply each feed to get the next higher feed,

$$f = \sqrt[n-1]{\frac{b}{a}}$$

in which b is the fastest, and a, the slowest speed of shaft P. That is, in the present case

$$R_p \text{ maximum} = 30 = b.$$
$$R_p \text{ minimum} = 0.6 = a.$$

The problem in our case stated that cones D and E, as well as pulleys X and Y could be transposed. The cones have four steps, and transposing them gives us eight speeds. Pulleys X and Y being also transposable gives, therefore, $2 \times 8 = 16$ speeds. The numerical value for f is therefore in our case,

$$f = \sqrt[15]{\frac{30}{0.6}} = \sqrt[15]{50}$$

The maximum and the minimum speeds of shaft P per one revolution of spindle of machine, as well as the number of steps required, being known, we now readily obtain a geometrical series with the minimum speed of shaft P as a beginning, and the maximum speed as the

last step. The numerical values that follow are found exactly in the same way as the values for the different speeds of a lathe drive as already shown. The required speeds of shaft P are then:

1) 0.6	5) 1.70	9) 4.83	13) 13.72
2) 0.78	6) 2.21	10) 6.27	14) 17.81
3) 1.01	7) 2.87	11) 8.14	15) 23.11
4) 1.31	8) 3.72	12) 10.57	16) 30.00

The value of p, in our case, becomes, according to formula (1),

$$p = \frac{R_p \times 0.25}{30} = 0.0083\, R_p$$

in which R_p, the number of revolutions of shaft P, has the different values found above. By substituting these values of R_p, we obtain the following feeds, which are the feeds of the lead screw per one turn of machine spindle.

1) $0.6 \times 0.0083 = 0.005$ inches	9) $4.83 \times 0.0083 = 0.0400$ inches
2) $0.78 \times 0.0083 = 0.0065$ "	10) $6.27 \times 0.0083 = 0.0520$ "
3) $1.01 \times 0.0083 = 0.0084$ "	11) $8.14 \times 0.0083 = 0.0677$ "
4) $1.31 \times 0.0083 = 0.0109$ "	12) $10.57 \times 0.0083 = 0.0877$ "
5) $1.70 \times 0.0083 = 0.0141$ "	13) $13.72 \times 0.0083 = 0.1138$ "
6) $2.21 \times 0.0083 = 0.0183$ "	14) $17.81 \times 0.0083 = 0.1513$ "
7) $2.87 \times 0.0083 = 0.0238$ "	15) $23.11 \times 0.0083 = 0.1918$ "
8) $3.72 \times 0.0083 = 0.0308$ "	16) $30.00 \times 0.0083 = 0.2500$ "

We now write the speeds found for shaft P in two columns, one containing the odd numbers and the other the even numbers, in this manner:

1) 0.6	2) 0.78
3) 1.01	4) 1.31
5) 1.70	6) 2.21
7) 2.87	8) 3.72
9) 4.83	10) 6.27
11) 8.14	12) 10.57
13) 13.72	14) 17.81
15) 23.11	16) 30.00

The series of speeds in each column forms a geometrical progression, and we assume that the speeds in the first column are due to the position of the pulleys X and Y as shown in the outline drawing, Fig. 1, and that the speeds in the second column are due to a reversed position of X and Y. That is to say, the speeds in the second column above are obtained after having changed Y to X and X to Y. As these speeds in the second column are equal to the speeds in the first column multiplied by factor f, it follows that the two speeds of shaft R are to each other as 1 is to f. Assuming these two speeds to be m and n, the proportion exists,

$$m : n = 1 : f \tag{3}$$

Supposing x and y to represent the diameters of the respective pulleys; it will be evident that

$$1 \times x = my\text{; or, } m = \frac{x}{y} \tag{4}$$

$$1 \times y = nx\text{; or, } n = \frac{y}{x} \tag{5}$$

Substituting the values (4) and (5) in formula (3) we have

$$\frac{x}{y} : \frac{y}{x} = 1 : f\text{, or } f = \frac{y}{x} : \frac{x}{y} = \frac{y}{x} \times \frac{y}{x} = \frac{y^2}{x^2} \tag{6}$$

The value of f being known, we have in formula (6) an expression of the relation which the diameters of the pulleys X and Y must bear to each other. Putting this formula into a more handy shape we find from $f = \frac{y^2}{x^2}$

$$y^2 = fx^2\text{, or } y = \sqrt{fx^2} \tag{7}$$

$$x^2 = \frac{y^2}{f}\text{, or } x = \sqrt{\frac{y^2}{f}} \tag{8}$$

In using either (7) or (8), and assuming one diameter, the other one is easily found. The remaining part of the problem, that is, to find the diameters of the cone, is now a simple matter.

CHAPTER III

MACHINE TOOL DRIVES

The present chapter contains considerable matter already treated in Chapter II. In order to make the present chapter a complete whole by itself, it has, however, been considered advisable to repeat such statements and formulas as are necessary to fully comprehend the somewhat different treatment of the subject presented in this chapter.

One of the first problems encountered in the design of a new machine tool is that of laying out the drive. The importance of a properly proportioned drive is coming more and more to be recognized. The use of high-speed steels, and the extra high pressure under which modern manufacturing is carried on, precludes the use of any but the most modern and efficient drive.

The drive selected may be one of the following different kinds, depending on the conditions surrounding the case in hand: We may make the drive to consist of cone pulleys only; we may use cone pulleys in conjunction with one or more sets of gears; or we may make our drive to consist of gears only, depending on one pulley, which runs at a constant speed, for our power. If the conditions will allow, we may use an electric motor, either independently or in connection with suitable gearing.

After having selected the form which our drive is to take and the amount of power to be delivered, which we will assume has been decided upon, we may turn our energies to the problem of arranging the successive speeds at which our machine is to be driven. As most machines requiring the kind of drive with which we are here concerned have spindles which either revolve the work, or a cutting tool that has to be worked at certain predetermined speeds dependent on the peripheral speed of the work or cutter, a natural question to be asked at this point is, "What is the law governing the progression of these speeds?"

As an example to show what relation these speeds must bear to one another, let us suppose that we have five pieces of work to turn in a lathe, their diameters being 1, 2, 5, 10, and 20 inches respectively. In order that the surface speed may be the same in each case we must revolve the one-inch piece twice as fast as the two-inch piece, because the circumference varies directly as the diameter, so that a two-inch piece would be twice as great in circumference as the one-inch piece. The five-inch piece would revolve only one-fifth as fast as the one-inch piece; the 10-inch piece 1/10th, the 20-inch piece 1/20th. We have seen that the addition of one inch to the diameter of the one-inch piece reduces the speed 100 per cent. If we add one inch to the two-inch piece we reduce the speed 50 per cent, and similarly one inch added to the 5-, 10-, and 20-inch pieces reduces the speed 20, 10, and 5 per cent respectively. From this we see that the speed must vary inversely with the diameter for any given surface speed. It also shows that the speeds differ by small increments at the slow speeds, the increment gradually increasing as the speed increases. Speeds laid out in accordance with the rules of geometrical progression fulfill the requirements of the above conditions.

If we multiply a number by a multiplier, then multiply the product by the same multiplier, and continue the operation a definite number of times, we have in the products obtained a series of numbers which are said to be in geometrical progression. Thus 1, 2, 4, 8, 16, 32, 64 are in geometrical progression, since each number is equal to the one preceding, multiplied by 2, which is called the ratio. The above may be expressed algebraically by the following formula:

$$b = a\,r^{n-1}$$

where b is a term or number which is the nth term from a which i the first term in the series. The term r is the ratio or constant multiplier.

If we are given the maximum and minimum of a range of speed: we may find the ratio by the following formula, when the number of speeds is given:

$$r = \sqrt[n-1]{\frac{b}{a}}$$

As most cases in which we would use this formula would require the use of logarithms, we will express the above as

$$\text{Log}\, r = \frac{\text{Log}\, b - \text{Log a}}{n - 1}$$

Let us suppose we are designing a drive which is to give a range of 18 spindle speeds, from 10 to 223 revolutions per minute. Now the first thing to be done is to find the ratio r, which, by the above formula is found to be 1.20, and by continued multiplication, the series is found to be 10, 12, 14.4, 17.25, 20.7, 24.85, 29.8, 35.8, 43, 51.6, 62, 74.4, 89.4, 107, 129, 155, 186, 223.

Our drive can be made to consist of one of the many forms just mentioned. As the cone and back-gear is the most common form, and fills the conditions well, we will choose that style drive for the case in hand. We may have a cone of six steps, double back-gears and one counter-shaft speed, such as would be used in lathe designs, or we may use a cone with three steps, double back-gears and two counter-shaft speeds as is used in milling machines. This latter plan will be followed in our present case.

There are two methods of arranging the counter-shaft speeds. First, by shifting the machine belt over the entire range of the cone before changing the counter-shaft speed; and second, by changing the counter-shaft speed after each shift of the machine belt. The method used will have a very important effect on the design of the cone. The cone resulting from the former practice will be quite "flat," with very small difference in the diameter of the steps, while the use of the second method will produce a cone which will have a steep incline of diameters. Some favor one, some the other. The controlling point in favor of the first method is the appearance of the cone obtained.

We will first design our drive with the conditions of the first method in view; that is, we will arrange our counter-shaft speeds so that the full range of the cone is covered before changing the counter-shaft speed, thus obtaining the flat cone. Tabulating the speeds in respect to the way they are obtained, we have

CONE.	Open Belt.		Small Ratio Back Gears in.		Large Ratio Back Gears in.	
	Fast Counter.	Slow Counter.	Fast Counter.	Slow Counter.	Fast Counter.	Slow Counter.
Step 1.....	223	129.	74.4	43.	24.85	14.4
Step 2.....	186	107.	62	35.8	20.7	12.
Step 3.....	155	89.4	51.6	29.8	17.25	10.
	1	2	3	4	5	6

From the above table we may obtain the ratio of the two sets of back-gears, the counter-shaft speeds, and the speeds off each step of the cone.

The ratio of the large ratio back-gears is found by dividing one term in column 2 by a corresponding term in column 6. The ratio of the small ratio gears is found by dividing a term in column 2 by a corre-

sponding term in column 4. The ratio of counter-shaft speeds is obtained by dividing a term in column 5 by a corresponding term in column 6; and the ratio of speeds off each step of the cone, by dividing the term corresponding to step 1 in any column by a term corresponding to step 2 or 3, as desired, from the same column. The results for the present case are as follows:

Ratio of large ratio gears is........................ 8.94 to 1
Ratio of small ratio gears is........................ 2.98 to 1
Ratio of counter-shaft speeds is.................... 1.725 to 1
Ratio of speeds off step 1 to those off step 2........ 1.2 to 1
Ratio of speeds off step 1 to those off step 3........ 1.44 to 1

The matter of designing the cone seems to cause trouble for a good many, if we are to judge by the results obtained, which are various in

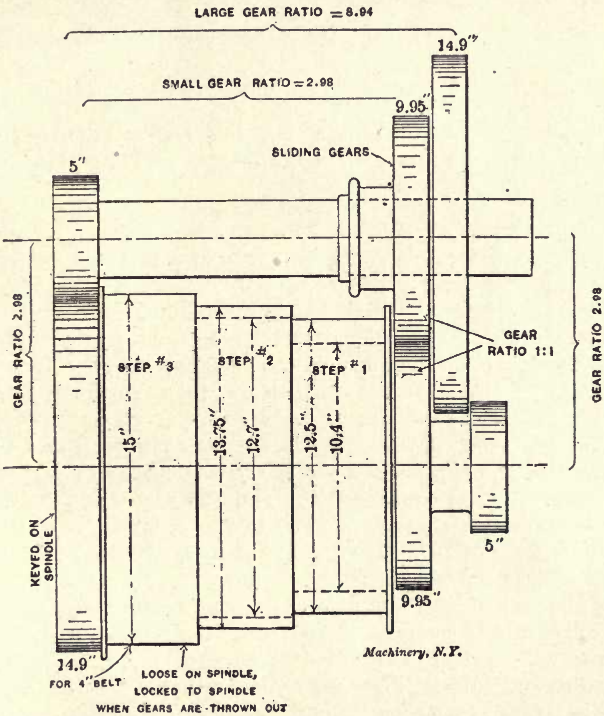


Fig. 2. Two Methods of Laying out the Cone for a Double Back-Geared Spindle

any collection of machine tools, even in those of modern design. It is possible to design a cone so as to obtain speeds in strict accordance with the geometrical series. In most cases the counter-shaft cone and the one on the machine are made from the same pattern, so that it is necessary that the diameters be the same for both cones, and since the belt is shifted from one step to another, its length must be kept

constant. This is accomplished by having the sum of diameters of corresponding steps equal.

We will take as the large diameter of the cone, 15 inches. The ratio of the speeds off step 1 and step 3 is 1.44 to 1. This ratio also equals $\frac{D \times D}{d \times d}$ where D is the diameter of largest step and d is the diameter of smallest step. Making them opposite terms in an equation we get,

$$1.44 = \frac{D \times D}{d \times d} = \frac{D^2}{d^2}$$

$$\text{or } 1.44 \times d^2 = D^2$$

$$d = \sqrt{\frac{D^2}{1.44}} = \sqrt{\frac{15 \times 15}{1.44}} = 12.5 \text{ inches, diameter of small step.}$$

The sum of the corresponding diameters on the cones is $15 + 12.5 = 27.5$.

Since this is a three-step cone the middle steps must be equal. Therefore $\frac{27.5}{2} = 13.75 =$ diameter of middle step. We found that the ratio of the speeds off first and second step is 1.2. Let us examine the above figures to see that the diameter of the middle step is correct. Thus,

$$\frac{15}{12.5} \times \frac{13.75}{13.75} = 1.2,$$

which is the correct ratio. This cone is shown in full lines in Fig. 2.

Let us now figure the diameter of the back-gears. We will assume that the smallest diameter possible for the small gears in the set is 5 inches. In order to keep the gears down as small as possible we will take this figure as the diameter of the small gear here. It is general practice, though obviously not compulsory, to make the two trains in a set of back gears equal as to ratio and diameters. When double back gears are used, the large ratio set is made with two trains of similar ratio. The small ratio set is then composed of two trains of gears whose ratios are unlike. The ratio of each train in the large ratic set, if taken as similar, is equal to the square root of the whole ratio; thus, in our drive we have $\sqrt{8.94} = 2.98$, and from this the large gear is $5 \times 2.98 = 14.9$ inches in diameter. The ratio of the small ratio set is equal to 2.98, and as one train of gears in the double back gear arrangement is common to both sets, the remaining train in the small ratio set must be of equal diameters, or $5 + 14.9 \div 2 = 9.95$ inches, as shown in Fig. 2. These figures will have to be slightly altered in order to adapt them to a standard pitch for the teeth, which part of the subject we will not deal with here.

In order to be able to compare the results of the two different methods of selecting counter-shaft speeds mentioned above, let us figure out the dimensions of a drive with counter-shaft speeds arranged according to the second method.

Proceeding in a manner similar to that pursued for the case treated above, we may tabulate the speeds as shown in the table on next page.

Cone.	Open Belt.		Small Ratio Gears in.		Large Ratio Gears in.	
	Fast Counter Speed.	Slow Counter Speed.	Fast Counter Speed.	Slow Counter Speed.	Fast Counter Speed.	Slow Counter Speed.
Step 1.....	223	186.	74.4	62.	24.85	20.7
Step 2.....	155	129.	51.6	43.	17 25	14.4
Step 3. ...	107	89.4	35.8	29.8	12.	10.
	1	2	3	4	5	6

The various ratios are:

Large ratio gears.................................... 8.94 to 1
Small ratio gears.................................... 2.98 to 1
Counter-shaft speeds.................................. 1.2 to 1
Speeds off step 1 to those off step 2.................. 1.44 to 1
Speeds off step 1 to those off step 3.................. 2.07 to 1

The cone dimensions are figured in the same manner as before and are 10.4 inches for step 1; 12.7 for step 2; 15 for step 3. This cone is shown dotted in Fig. 2.

We are now in a position to compare the results given by the two methods above referred to. Let us make the first comparison from the point of view of power delivered by the belt. It is well-known that the power of a belt is directly proportional to the speed at which it runs. This fact gives us an easy means of comparing our two designs. We will do this by charting the speed in feet per minute of the belt when running on the different steps of the two cones for each spindle speed. This has been done in Fig. 3, where the full lines show the curve for the first method, and the dotted lines show that for the second method. The curves at the left are those for the slow counter speeds, while at the right are seen those for the fast counter speeds. Attention is called to the great difference in power delivered between the two counter speeds in the first case, while the two sets of curves for the second method lie close together. Also, note the gain in power at speeds obtained through the slow counter in the second case. The power lost in the second case on the fast counter speeds will not be felt so much, for the same principle applies here as it does to the strength of beams, bridges, etc., *viz.*, a chain is no stronger than its weakest link.

The constant-speed pulley drive has become quite a common feature in machine tool design, and has become quite a strong favorite with many. Had our machine been provided with a drive of this design, we would have had a curve on the chart as shown by the vertical full line. The power delivered by the belt would have been constant throughout the full range of speed. This curve also applies to the motor drive, when a constant-speed motor, or a variable-speed motor of the field control type, is used, although slight modifications would have to be made for the decrease in efficiency at the extremes of the

speed range of the latter type motor, which would cause a slight bend in the curve, making it convex toward the right. Motors using the multiple-voltage system, or the obsolete armature resistance control, would show curves quite as irregular as those from the cone and back-gear drive.

Another method of comparison is by charting the pull or torque at the spindle for each spindle speed. This is done in Fig. 4, where the

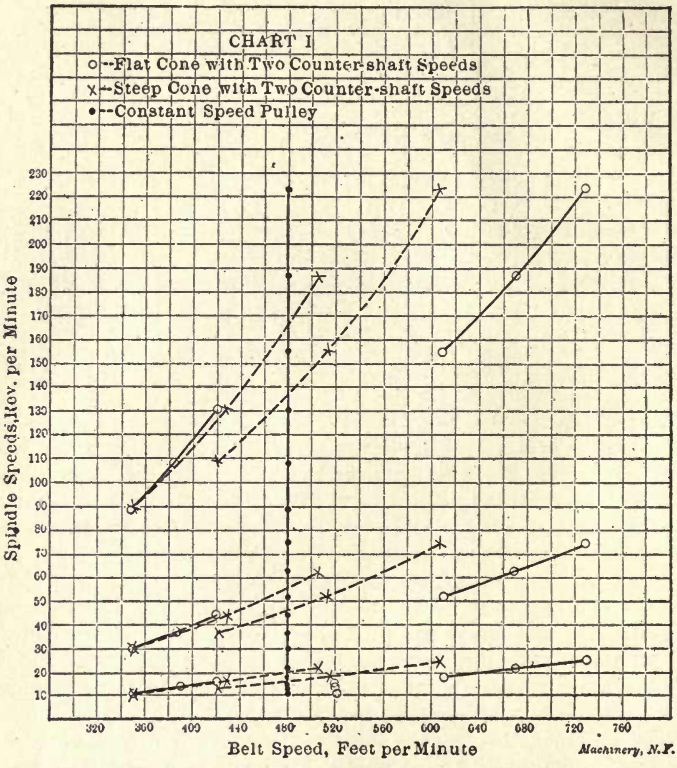


Fig. 3. Variation in Belt Speeds for Various Methods of Driving

constant speed pulley drive is shown by the full line, and is used as a comparator by which to compare the results of the two drives treated above. This figure is self-explanatory and will not need to be interpreted, but attention may be called to how much better the drive of the second case follows the ideal line than does that of the first method. This chart also shows how very close a cone and double back-gear drive comes to the constant belt-speed drive with equal power at all speeds.

Much has been said about the relative values of the two styles of cone pulleys treated above, but the charts given herewith will no doubt surprise some, and may be the means of turning them in favor

of the second method. The only good point the first method has over the second is in the appearance of the cone which has, apparently, powerful lines, which are, however, misleading, as has been shown.

Another disadvantage of the first method is the wide ratio of the

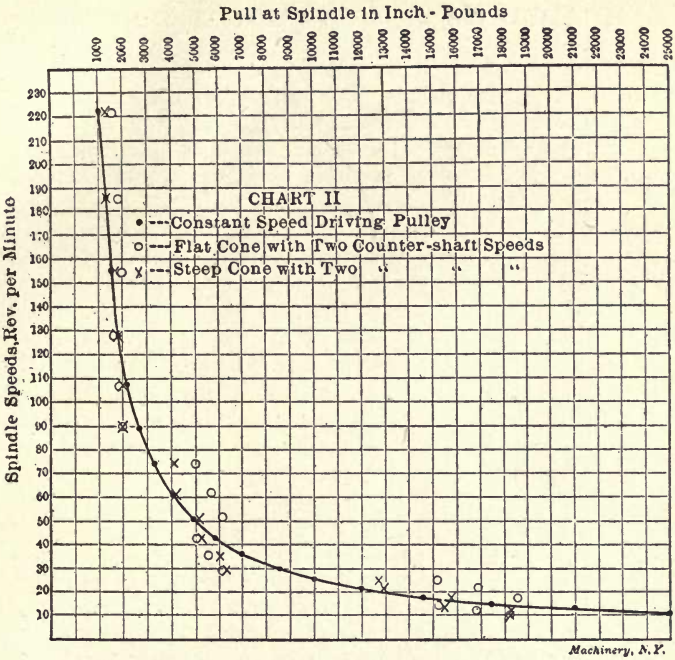


Fig. 4. Comparison of Torques for Various Methods of Driving

counter-shaft speeds, where, in order to get sufficient power out of the slow speed counter-shaft belt, we must have the high-speed pulley running at almost prohibitive speed, which soon tells, and as loose pulleys are a source of annoyance when their speed is moderate, trouble is sure to appear when the limit of speed is approached.

CHAPTER IV

GEARED OR SINGLE PULLEY DRIVES

Whether the geared drive, so called in order to distinguish it from the belt drive used with stepped cone pulleys, originated with some machine tool builder who was desirous of improving a given machine, or whether it was first suggested by a machine tool user in an endeavor to secure better facilities for machine operation, would be interesting to know, but difficult to determine.

Whatever the origin, the geared drive is a response to a demand for a better method of speed variation than could be obtained from stepped pulleys and a movable belt. The gradually growing demand for more powerful machine drives in the past has led to the widening of belts to the maximum point consistent with a desirable number of steps of the pulley, and the ease of belt shifting. The limiting point for belt width may be said to be reached when a belt can no longer be shifted easily by hand. For some machines, notably lathes, the maximum diameters of the driving pulleys are generally limited by conditions inherent in the machine themselves.

Back-gears were in many instances increased in ratio to make up for what could not be had by further increase of belt widths or pulley diameters, until in some cases the gap between speeds obtained directly by the belt and those obtained through the back gears became too great. When such conditions were reached, obviously, the next suggestion involved the combination of a constant speed belt of such a width and operated at such a speed as to give the requisite power, in connection with some combination of gears to be used for obtaining the desired variation in speeds. Such a combination is, in fact, a reversion of type; a going back to a system of driving formerly much used by foreign builders of machine tools. Many foreign builders objected to the use of stepped pulleys, considering their use as a deviation from, or, as being contrary to, good mechanical practice, preferring in many cases to secure speed variation by means of separate changeable gears. The objectionable feature of such a system did not suit American ideas, hence the early adoption of stepped pulleys and a movable belt as a means of quickly effecting changes even though the device was and is still considered by some designers as anomalous or paradoxical from the standpoint of pure mechanics. The substitution of the variable speed geared drive for the stepped pulley drive is therefore not due to any inherent defect in the stepped pulley so much as to its limitations as previously mentioned, and to a desire for improved facilities for quickly obtaining speed variations.

For belt-driven machines that require a variable speed, the geared drive will probably come more into use whenever its adoption will be justified from a productive or a commercial standpoint. Whatever

defects may be existent in any of its varied forms will be tolerated just as long as it meets and fulfills required conditions.

As a device of utility the geared drive has passed the point where it might by some have been considered as a fad. As a matter of fact, scarcely any new device representing a radical departure from generally accepted design and practice has ever been brought out that was not considered a fad by some one. The history of machine tool progress has shown that the fad of yesterday has frequently become the custom or necessity of to-day. Extreme conservatism will see a fad where progress views an undeveloped success. One drawback to the general adoption of any geared drive is its cost, and this will determine in most cases whether it or a belt drive shall be used; it is a matter requiring careful judgment to determine the point where the results obtained justify the added expense.

It is, however, with very few exceptions, the opinion among builders and users of machine tools that the single pulley drive will largely supersede the cone drive. Still for certain conditions it is doubtful whether we will find anything better than our old servant, the cone. The two principal advantages possessed by the single pulley drive are: First, a great increase in the power that can be delivered to the cutting tool owing to the high initial belt speed. The belt speed always being constant, the power is practically the same when running on high or low speeds. The cone acts inversely in this respect; that is, as the diameter of the work increases, for a given cutting speed, the power decreases. As a second advantage, the speed changes being made with levers, any speed can be quickly obtained.

To these might be added several other advantages. The tool can be belted direct from the lineshaft; no counter-shaft is required; floor space can be economized. It gives longer life to the driving belt; cone belts are comparatively short-lived, especially when working to their full capacity. There are, however, some disadvantages to be encountered. Any device of this nature, where all the speed changes are obtained through gears, is bound to be more or less complicated. The first cost, as mentioned, is greater. There is also more waste of power through friction losses. A geared drive requires more attention, break-downs are liable to occur, and for some classes of work it cannot furnish the smooth drive obtained with the cone. Most of these objections, however, should be offset by the increased production obtained.

To the designer the problem presented is one of obtaining an ideal variable speed device, something that mechanics have been seeking for years with but poor success, and it is doubtful whether we will get anything as good for this purpose as the variable speed motor in combination with double friction back-gears and a friction head. There are, it is true, some very creditable all-gear drives on the market in which the problem has been attacked in various ways. Still there is ample room for something better. The ideal single pulley drive should embody the following conditions.

1. There should be sufficient speed changes to divide the total range

into increments of say between 10 and 15 per cent.

2. The entire range of speeds should be obtained without stopping the machine.

3. Any speed desired should be obtained without making all the intermediate changes between the present and desired speed.

4. All the speeds should be obtained within the tool itself, and no auxiliary counter-shaft or speed variators should be used.

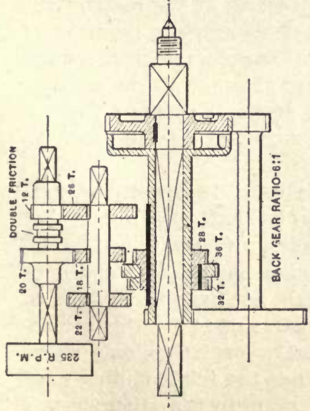
5. Only the gears through which the speed is actually being obtained should be engaged at one time.

6. The least possible number of shafts, gears and levers should be used.

There are few subjects in machine design which admit of so many combinations, arrangements and devices. In Figs. 5 to 10, inclusive, are shown some examples taken at random from a large collection. All of these, except Fig. 10, have the number of teeth and the speeds marked. Each has some good points, but none of them possesses all the points referred to above. The only reason for showing them is to show what a vast number of designs can be devised. One of them, that shown in Fig. 5, has been built, a number of machines have been running for over a year, and they give very good results. In Fig. 11 is shown the way the idea was worked out, as applied to a 20-inch Le Blond lathe.

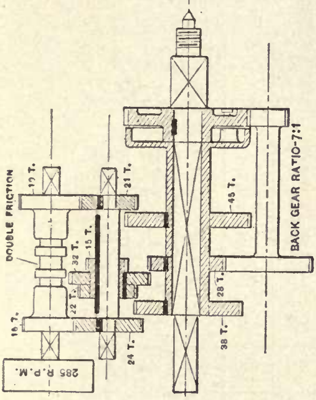
The design for the headstock shown in Fig. 11 needs little explanation since the drawing shows the parts quite clearly. The friction clutch on the driving-shaft *Z*, which alternately engages pinions *H* and *J*, is of the familiar type used in the Le Blond double back-geared milling machine. Sliding collar *D*, operated by handle *S*, moves the double tapered key *E* either to the right or left as may be desired, raising either wedge *W* or *W'*, which in turn expand rings *X* or *Y* within the recess in either of the two cups, *F* and *F'*. Either of two rates of speed is thus given to quill gear *K* and the two gears *L* and *M* keyed to it. On the spindle is a triple sliding gear which may be moved to engage *P* with *M*, *O* with *L* (as shown in the drawing) or *N* with *K*, thus giving three changes of speed when operated by lever *T*. The six speeds obtained by the manipulation of levers *S* and *T* are doubled by throwing in the back-gears, giving 12 speeds in all.

In comparing the merits of a series of gear drive arrangements like those shown in Figs. 5 to 10, one might apply the "point" system in determining the most suitable one. The number of points that are to be assigned to a device for perfectly fulfilling any one of the various requirements would be a matter requiring nice discrimination. So the method outlined below is to be taken as being suggestive, rather than authoritative. The first requirement is that there shall be sufficient speed changes to divide the total range into increments of between 10 and 15 per cent. The six schemes proposed do not all, unfortunately for our proposal, take in the same range of speed; considering, however, that they were each to be designed to give from 9 to 240 revolutions per minute to the spindle, as in case Fig. 5, and that a 15 per cent increment is to be allowed, the number of changes required



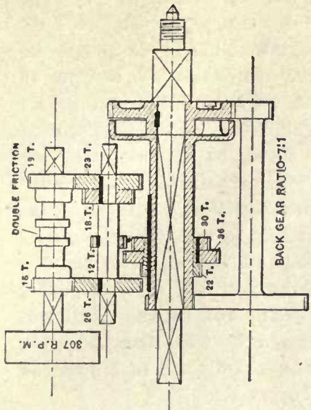
SPINDLE SPEEDS
B. GEARS IN-9.-12.4-16.7-21.6-29.76-40
B. GEARS OUT-54.-74.-100.-129-178-240

Fig. 5



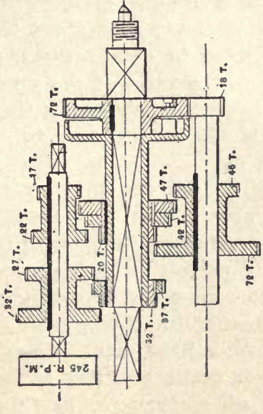
SPINDLE SPEEDS
9-12-16-24-32-48
63-84-112-168-224-336

Fig. 6



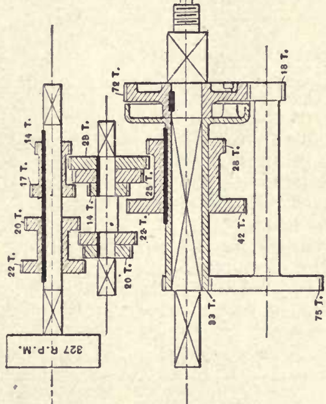
SPINDLE SPEEDS
9-12-16-24-32-48
63-84-112-168-224-336

Fig. 7



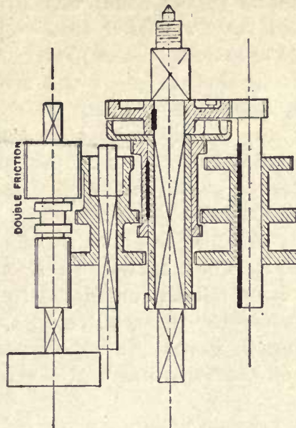
SPINDLE SPEEDS
D. GEARS OUT-6-8-11-16
1ST B. GEARS IN- 22-32-45-62
2ND B. GEARS IN-89-128-179-245

Fig. 8



SPINDLE SPEEDS
B. GEARS IN-6-8-11-14-18-25-33 40
B. GEARS OUT-54-72-95-136-164-222-297-359

Fig. 9



SPINDLE SPEEDS
24 SPEEDS OBTAINABLE
NO DATA GIVEN

Fig. 10

can be found in the usual way by dividing the logarithm of 27— the total speed ratio required (240 ÷ 9 = 27)—by the logarithm of 1.15, which is the ratio of the geometric series desired. This gives 24 speeds, about, as needed to meet the requirements. Suppose we assign 15 points to a machine having 24 speeds. Let us set this down in its proper place in the suggested table given below. For the second qualification, that the machine shall not have to be stopped, we may assign 20 points to the ideal machine. The principle of "selective" control is assigned 10 points. The fourth consideration, requiring that all speeds shall be obtained within the tool itself is a positive requirement. If it is not met, the mechanism is out of the contest, so this question need not be considered in our table of points. Fifteen points are suggested for the requirement that the gears not in use shall not be running in mesh. The sixth requirement reads "The least possible number of shafts, gears and levers should be used." It is suggested

A SUGGESTED TABULATION OF THE MERITS OF THE VARIOUS DRIVES PROPOSED

Requirements.	Perfect Design.	No. 1	No. 2	No. 3	No. 4	No. 5	No. 6
No. of changes required compared with No. obtained..	15	8	8	8	8	10	15
Stopping of machine	20	14	14	14	14	14	14
"Selective" control........	10	7	7	7	7	7	7
Gears not in use, must not be in mesh	15	13	13	13	15	15	13
Ratio of No. of changes to No. of movements	20	15	15	15	13	12	14
Ratio of No. of changes to No. of gears............	20	10	9	9	9	16	18
Total..................	100	67	66	66	66	74	81

that this be divided, giving 20 points to the question of the ratio of the number of changes obtained to the number of movements required of the operator to obtain them, and giving the same number of points to express the ratio of the number of changes obtained to the number of gears used in obtaining them. The sum of these points added together is 100, which may be considered as representing the ideal design.

In filling out the table, since Fig. 5 has only 12 speeds or half the number required, we will give it only one-half the number of points, dealing similarly with the other designs up to No. 6, in Fig. 10, which is perfect in this respect. The machine has to be stopped to throw in back-gears. Assuming that this would not have to be done in 70 per cent of the changes, we get a uniform value of 14 for this consideration for all the cases. The feature of selective control is only about two-thirds realized in any of these designs, since the triple sliding gear used in all of them, in moving from one extreme to the other, passes through an intermediate position which is not required at the time.

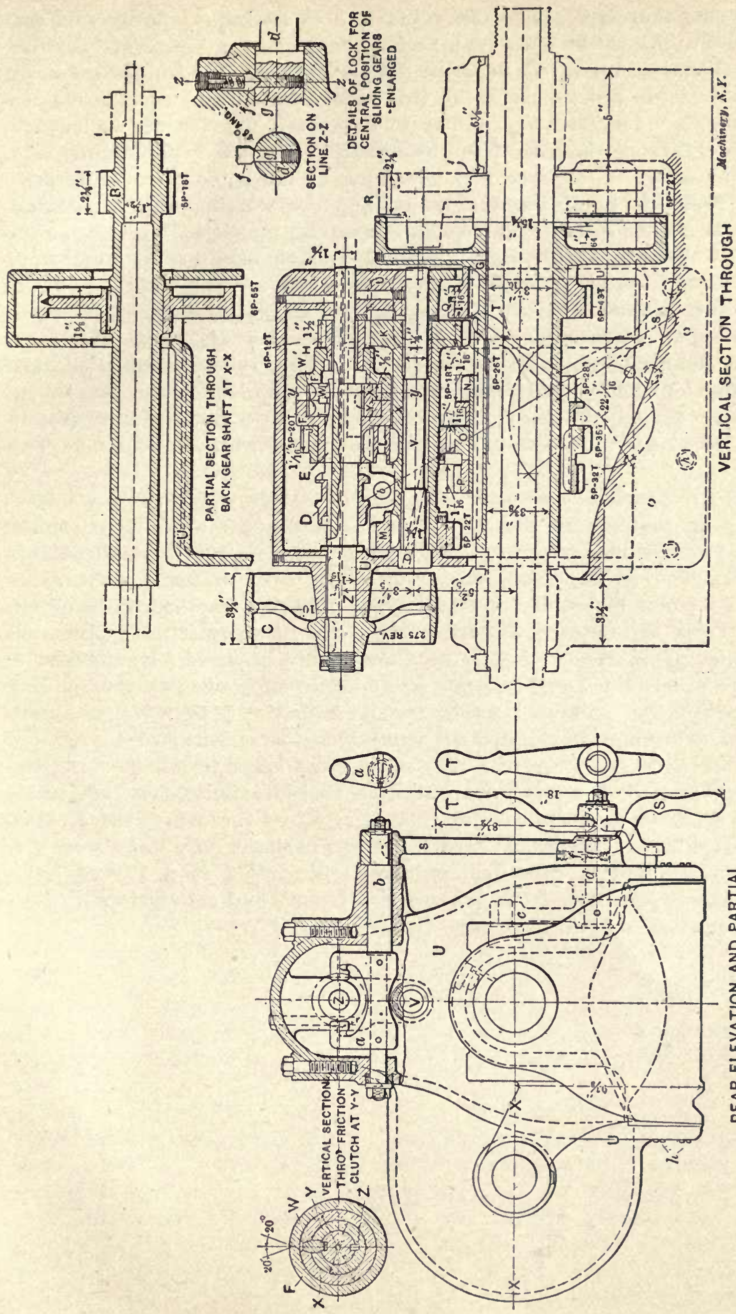


Fig. 11. The Layout shown in Fig. 5 applied to Head-stock of a 20-inch Lathe

We may therefore assign the value 7 to each of these designs on this account. As to the question whether the gears not in use are running idly in mesh, all the designs are nearly perfect. The values set down in this table are suggested by this consideration. In considering the number of movements required to effect the number of changes obtained, the throwing in of the back-gear is credited with four motions, the stopping of the machine, unlocking of the spindle from the gear, the throwing in of the back-gears, and the starting of the machine. The 20 points of the ideal machine are then multiplied by each of the ratios obtained by dividing the number of changes by the number of movements, and the number of points found are set down as shown. For the last item, twice as many changes as there are gears employed is taken as a maximum which can probably not be exceeded. With this as a standard, the ratio obtained by dividing the number of changes by the number of gears used is employed to calculate the number of points. Adding the number of points obtained in each column we find that No. 1 has 67, No. 2, 3, and 4 each have 66, while No. 5 has 74, and No. 6, 81.

The comparison has been undertaken in this way with the understanding that all the arrangements are susceptible of being embodied in a practicable design. That arrangement No. 6 is practicable is strongly to be doubted. The number of teeth in the various gears used are not given, and it is far from probable that one could obtain with this arrangement a series of speeds in geometrical progression by moving in regular order the three levers required. Nos. 4 and 5, while otherwise well arranged, are open to the objection that sliding gears rotating at high rates of speed are used. This, if valid, constituted a disqualifying objection similar to that mentioned in relation to the fourth requirement. The first three cases in which a friction clutch instead of sliding gears is used on the driving shaft are therefore much to be preferred for this reason. Of these first three cases, our tabulation shows that case No. 1 has a slight advantage, and Fig. 11, in which this arrangement has been applied to a 20-inch lathe headstock, shows that the scheme is a simple and satisfactory one, so far, at least, as one can judge from a drawing.

CHAPTER V

DRIVES FOR HIGH-SPEED CUTTING TOOLS

What has been considered in the past as marvelous in the performance of high-duty cutting tools may now be compared with the proved results of air-hardening cutting tools. The metallurgist has proved to us, and a great many machine tool builders have satisfied themselves by practical experiment, that the high-speed cutting steels are at our service, but they must be properly shod if they are to be used to the best advantage. Some concerns who have experimented with the high-speed steels, and who anticipated much, have failed through lack of a proper analysis of the conditions which accompany the use of the high-speed cutting steels. It takes but a moment's reflection to convince one of the absurdity of trying to get as effective a fire from a six-inch as from a thirteen-inch gun, even though the same explosive charge is used in both.

Some viewed this unusual commotion about the high-speed cutting steels as being somewhat fanatical or a fad which would rage for a time, and then die a natural death, as many others have done. True, this was not the first high-duty cutting steel which had been advanced with enormous claims of efficiency. Mushet steel had been on the market for several years, and the great things predicted for it did not fully meet everybody's expectations. The chief reason for this was its far too limited use in a great many cases, on account of its being expensive, difficult to forge, grind, and to get a satisfactorily finished surface with it, and the failure of the machine to stand up to the chip it could take. Then again, when Mushet steel was introduced, competition among machine tool builders for increased product from their machines did not begin to compare with that which now exists with firms which more than ever are on an intensely manufacturing basis. Manufacturing plants of any considerable size using metal cutting tools are bidding nowadays for special machinery of the simplest form to augment the output of a single product, and not comparatively complicated combination tools, designed for many operations on many pieces, and which save considerable room and first cost of installation, but are of necessity inconvenient, and unsuitable for high-duty service.

The complaint which has been made by some that the new high-speed cutting steels are unfit for finishing surfaces cannot be consistently sustained. The modernly-designed manufacturing grinder has unquestionably proved to be the proper tool for finishing surfaces from the rough; and undoubtedly, and beyond peradventure, the grinder is the natural running mate for the high-duty turning lathe and planer; and it seems probable that, instead of the grinder being a rarity and a luxury in shops, as a sort of tool-room machine, it

will be as much in evidence for manufacturing purposes as the more commonly-known machine tools of the present, or more so.

The innovations of the day in machine tool evolution are in most remarkable harmony and synchronism. The electric motor, which is fast developing the independent machine drive, demands a high speed for maximum efficiency of the motor; and what do we find contemporaneously developed but the high-speed cutting steels, the practicable commercial grinder, and the comparatively high-speed non-stroke milling machine to supersede the comparatively slow multi-stroke planer? Unquestionably, there never has been in the whole history of the machine tool business such an opportunity for the enterprising capitalist, the engineer, and the designer, to invest their money, brains and skill in a type of machine tools that will be as different from the present type of machine tools as the nineteentth century lathe is from the simple and crude Egyptian lathe of tradition.

The development of the cutting or producing end of the machine appears to be further advanced than the driving end. The direct motor drive without inter-connecting belts, chains, and gears is undoubtedly the simplest, most convenient, and most effective. The motor which is most desired has not been designed, but it should be a comparatively slow-speed motor having high efficiency, whose speeds vary by infinitesimal steps between its minimum and maximum limits, fully as simple as the "commutatorless" type, and with far higher pressures than are now used. In the meantime, during the process of development, we shall have to be content with the usual compounding elements between the motor and the driving spindle; but these compounding elements, in order to keep up with the procession, will naturally undergo revolutionary changes in design.

The silent chain drive and the high-speed motor are mutual helpmates; geared variable speed devices and single-speed induction motors are well wedded, but cone pulleys are practically just beginning to receive that examination and attention which can fit them for the service of higher speeds.

In the case of a turning lathe, as would naturally be expected, we are very much limited in the range of the sizes of pieces that can be turned—if we maintain an efficient range of speeds and sufficient diameters and widths of pulleys for surface speeds of belts—unless we use an abnormally ponderous cone pulley, which is entirely out of the question. To make this point clear, it may be well to analyze a specific case. We will assume that the lathe is designed with a four-stepped cone and with "front-gears" (the speed ratios of front-gears are figured the same as back-gears, but their thrust at the front box is opposite in direction to that of the back-gears and to the lifting effect of the tool, as it properly should be), two countershaft speeds, and for cutting 30-point carbon steel at a speed of 100 feet per minute with a chip of 5/16 by 3/32 inch cross section. It is furthermore assumed that the work and cutting tool are rigidly supported, and that the cutting tool has the proper amount of rake for least resistance and a fair amount of endurance.

Calculation of Cutting Force of Tool, and Speed of Belt

In order to make absolute computations of the required diameters, we should have reliable data on the amount of cutting force at the cutting edge of the tool when cutting the various metals at high speeds, reliable data for the best efficiency of the redesigned machine, and the approximate distance between the centers of the driving spindle and counter-shaft. Several experiments were made by Hartig, and subsequently by others, on the horse-power required at the cutting edge of a tool when cutting various metals at slow speeds with the ordinary tempered steels. The horse-power was determined by multiplying the weight of chips turned off per hour by a constant whose value varied with the degree of hardness of the metal cut and the conditions of the cutting edge of the tool. The average of the several constants for about 30-point carbon steel seems to be about 0.035.

Hartig's expression is given in the formula

$$H.P. = cW = 0.035 \times \pi \times D \times n \times d \times t \times 0.28 \times 60 \qquad (9)$$

and the usual expression for horse power is given in the form,

$$H.P = \frac{FS}{33000} = \frac{F \times \pi \times D \times n}{33000 \times 12} \qquad (10)$$

in which

$H.P.$ = horse power absorbed at the cutting edge of tool.
c = constant 0.035.
W = weight of chips per hour.
D = mean diameter of the area turned off per hour.
n = revolutions per minute.
d = depth of chip.
t = thickness of chip.
0.28 = assumed average weight per cubic inch of 30-point carbon steel.
F = force at cutting edge of tool.
S = distance through which force F acts.

Equating (9) and (10),

$$F = 0.035 \times 0.28 \times 60 \times 33000 \times 12 \times d \times t = 232850\ dt.$$

Since the chip assumed to be cut is 5/16 by 3/32 inch cross section, then the force at the cutting tool is

$$F = 232850 \times 5/16 \times 3/32 \text{ inch} = 6820 \text{ pounds.}$$

If the cutting speed is 100 feet per minute then the work at the tool

$$W = 6820 \times 100 = 682000 \text{ foot-pounds.}$$

If the efficiency of the machine is assumed at 85 per cent, then the effective work of the belt must be

$$W = \frac{682000 \times 100}{85} = 802500 \text{ foot-pounds.}$$

We will assume that a 5-inch double belt is the practical limit for the belt which can be conveniently used on the machine, and that the effective pull is 70 pounds per inch width when wrapped around a cast-

iron pulley with a contact surface of 180 degrees. The total effective pull is then

$$5 \times 70 = 350 \text{ pounds.}$$

Since our belt must deliver 802500 foot-pounds per minute, its velocity will be

$$V = \frac{802500}{350} = 2295 \text{ feet per minute, approximately,}$$

which must be proportional to the diameters of the cone pulleys and the counter-shaft speeds, which are obtained as follows.

It is customary to consider speeds in a series of geometrical progression if the most efficient and convenient range of speeds is desired. The constant multiplier will then be

$$r = \left(\frac{l}{a}\right)^{\frac{1}{n_1 - 1}}, \text{ in which} \qquad (11)$$

$r =$ constant multiplier.
$l =$ maximum R. P. M. of spindle.
$a =$ minimum R. P. M. of spindle.
$n_1 =$ number of speeds.

Let it be assumed that the lathe is designed to turn sizes from 1 to 6 inches. The corresponding maximum and minimum revolutions per minute for the cutting speed 100 feet per minute are 382 and 62, approximately. Then from (11)

$$r = \left(\frac{382}{62}\right)^{\frac{1}{15}}$$

$$\log r = \frac{1}{15} \log 6.16$$

$$r = 1.128$$

The whole series of speeds in geometrical progression and the diameters of stock, which will approximately correspond, if a cutting speed of 100 feet per minute be used, is given in the following table:

SPEEDS IN R.P.M.	DIAMETER OF STOCK			SPEEDS IN R.P.M.	DIAMETER OF STOCK		
382	1	CONE PULLEY SPEEDS	FAST COUNTERSHAFT SPEED	145	$2\frac{5}{8}$	CONE PULLEY SPEEDS	SLOW COUNTERSHAFT SPEED
338	$1\frac{1}{8}$			128	3		
300	$1\frac{1}{4}$			113	$3\frac{3}{8}$		
265	$1\frac{1}{2}$			101	$3\frac{3}{4}$		
235	$1\frac{5}{8}$	BACK GEAR SPEEDS		89	$4\frac{5}{16}$	BACK GEAR SPEEDS	
208	$1\frac{13}{16}$			78	$4\frac{7}{8}$		
184	$2\frac{1}{16}$			69	$5\frac{1}{2}$		
163	$2\frac{3}{8}$			62	6		

Industrial Press, N.Y.

In Fig. 12, assume that the counter-shaft and spindle cone pulleys are the same size, as is usually the case for the engine lathe. Let

$D_1 =$ diameter of largest step.
$D_4 =$ diameter of smallest step.
$n' =$ slowest speed of countershaft.

N_1 = fastest speed of spindle to correspond with slowest countershaft speed.

N_4 = slowest speed of spindle without back-gears to correspond with slowest countershaft speed.

$$\text{Let } \frac{D_1}{D_4} = r \qquad (12)$$

$$\frac{D_4}{D_1} = \frac{1}{r} \qquad (13)$$

Then

$$n' \times r = N_1 \qquad (14)$$

$$n' \times \frac{1}{r} = N_4 \qquad (15)$$

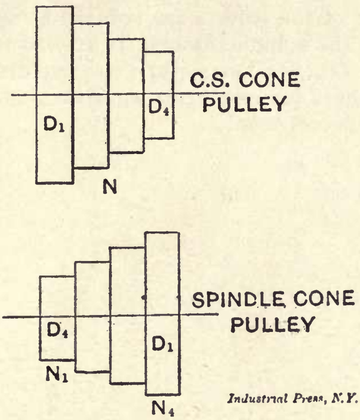


Fig. 12

Combining (14) and (15),

$$n' = \sqrt{N_1 \times N_4} \qquad (16)$$

Substituting in (16) the proper speeds taken from the table;

$$n' = \sqrt{145 \times 101} = 121$$

From (14)

$$r = \frac{N_1}{n'} = \frac{145}{121} = 1.199$$

$$D_4 = \frac{V}{\pi n'} \qquad (17)$$

Substituting in (17) the value of V and n',

$$D_4 = \frac{2295 \times 12}{3.14 \times 121} = 72\tfrac{1}{2} \text{ inches.}$$

From (12)

$$D_1 = r \times D_4 \qquad (18)$$

Substituting in (18) the value of r and D_4

$$D_1 = 1.199 \times 72\tfrac{1}{2} = 87 \text{ inches.}$$

The front gear ratio from spindle cone speed to driving spindle speed will be $\frac{145}{89} = 1.629$.

Since the values of the constants used in computing the force at the cutting tool were taken from experiments made with slow cutting speeds, and would be considered low in view of the fact, noted by some, that the work at the tool for high speeds increases in far greater proportion than the increased cutting speeds; and since the assumed 70 pounds per inch width for effective pull at the belt is quite liberal, it is clear that the pulleys are practically at a minimum size under the conditions assumed. It is therefore convincingly apparent that for the ordinary back-geared head, belts can be of no avail for high-speed cutting except for extremely limited ranges of diameters of stock.

If the diameters of the pulleys are reduced by speeding up the belts and gearing down the spindle, nothing is availed in most cases but an added and useless expense, since every compounding element is a loan for a mortgage whose interest rates sometimes increase pretty nearly in a geometrical progression.

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